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(54) **Method and apparatus for automatic clutch control with pid regulation.**

(57) An automatic clutch controller for a vehicle that reduces the oscillatory response to clutch engagement. The automatic clutch controller receives inputs from an engine speed sensor (13) and a transmission input speed sensor (31) and develops a clutch actuation signal controlling a clutch actuator (27) from disengaged to fully engaged. The clutch engagement signal at least partially engages the friction clutch (20) in a manner to cause the measured transmission input speed to asymptotically approach engine speed employing an approximate inverse model of this oscillatory response. The automatic clutch controller preferably includes a PID function and a differential engine speed function, which together adaptively adjust clutch engagement corresponding to vehicle loading. The automatic clutch controller includes a PID regulator (65) responsive to the difference of engine speed and transmission input speed, a prefilter (68) operating on the resulting PID signal, and a compensator (70) constructed to reduce the need for detailed particularization for individual vehicles or vehicle models by reducing the system closed loop sensitivity to vehicle parameter variations.

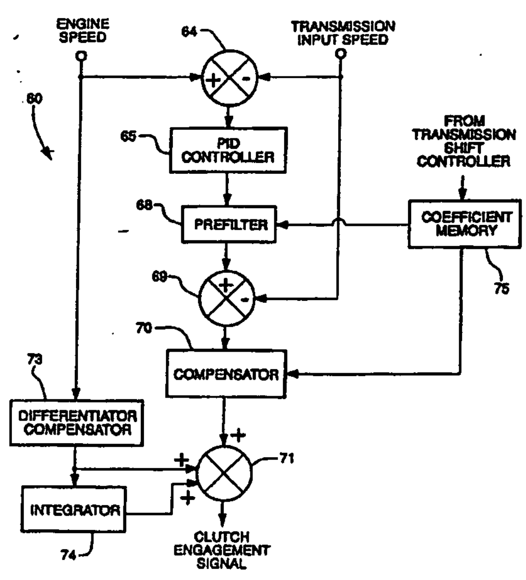

FIG - 4

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Technical Field of the Invention

The technical field of this invention is that of automatic clutch controls, and more particularly closed loop automatic clutch control and method for reducing oscillatory response to launch of a motor vehicle.

Background of the Invention

In recent years there has been a growing interest in increased automation in the control of the drive train of motor vehicles, and most especially in control of the drive train of large trucks. The use of automatic transmissions in passenger automobiles and light trucks is well known. The typical automatic transmission in such a vehicle employs a fluid torque converter and hydraulically actuated gears for selecting the final drive ratio between the engine shaft and the drive wheels. This gear selection is based upon engine speed, vehicle speed and the like. It is well known that such automatic transmissions reduce the effectiveness of the transmission of power from the engine to the drive shaft, with the consummate reduction in fuel economy and power as compared with the skilled operation of a manual transmission. Such hydraulic automatic transmissions have not achieved wide spread use in large motor trucks because of the reduction in efficiency of the operation of the vehicle.

One of the reasons for the loss of efficiency when employing a hydraulic automatic transmission is loss occurring in the fluid torque converter. A typical fluid torque converter exhibits slippage and consequent loss of torque and power in all modes. It is known in the art to provide lockup torque converters that provide a direct link between the input shaft and the output shaft of the transmission above certain engine speeds. This technique provides adequate torque transfer efficiency when engaged, however, this technique provides no gain in efficiency at lower speeds.

It has been proposed to eliminate the inefficiencies inherent in a hydraulic torque converter by substitution of an automatically actuated friction clutch. This substitution introduces another problem not exhibited in the use of the hydraulic torque converters. The mechanical drive train of a motor vehicle typically exhibits considerable torsional compliance in the driveline between the transmission and the traction wheels of the vehicle. This torsional compliance may be found in the drive shaft between the transmission and the differential or the axle shaft between the differential and the driven wheels. It is often the case that independent design criteria encourages or requires this driveline to exhibit considerable torsional compliance. The existence of substantial torsional compliance in the driveline of the motor vehicle causes oscillatory response to clutch engagement. These oscillatory responses can cause considerable additional wear to the drive train components and other parts of the vehicle. In addition, these oscillatory responses can cause objectionable passenger compartment vibrations.

The oscillatory response of the driveline to clutch engagement is dependent in large degree on the manner in which the input speed of the transmission, i.e. the speed of the clutch, approaches the engine speed. A smooth approach of these speeds, such as via a decaying exponential function, imparts no torque transients on clutch lockup. If these speeds approach abruptly, then a torque transient is transmitted to the driveline resulting in an oscillatory response in the vehicle driveline. United States patent application S.N. 772,204, filed October 7, 1991 and entitled "CLOSED LOOP LAUNCH AND CREEP CONTROL FOR AUTOMATIC CLUTCH", teaches the minimization or elimination of torsional oscillations due to compliance in the driveline during clutch engagement by controlling the clutch actuation to effect a smooth engagement. Subsequent patent applications, listed here, are improvements which make the control more robust. My previous patent application entitled CLOSED LOOP LAUNCH AND CREEP CONTROL FOR AUTOMATIC CLUTCH WITH ROBUST ALGORITHM, now patent number U.S. 5,275,267, addresses the same problem and includes a prefilter to shape the system transient response and reduces the need for detailed particularization for individual vehicles or vehicle models. United States patent application S.N. 08/165,957, filed Dec. 14, 1993 and entitled "METHOD AND APPARATUS FOR ROBUST AUTOMATIC CLUTCH CONTROL" is based on the same system and further improves robustness by overcoming the possibility of engine overload imposed by aggressive clutch engagement under certain conditions which leads to engine speed droop and even clutch dumping to avoid stalling the engine. The system as disclosed in the above specifications includes a slip integrator or actually two integrators in series which have the potential of being too sensitive to inner loop variations, leading to difficulty of control under some circumstances.

Thus it would be an advantage to provide automatic clutch actuation of a friction clutch that reduces the sensitivity to inner loop variations and offers better control. This invention is based in part on that previous work and adds additional robustness. The robustness permits the mass manufacture of transmissions applicable to a wide range of heavy duty trucks without individual tuning for a given truck type or load range.

Summary of the Invention

This invention is an automatic clutch controller used in a combination including a source of motive power, a friction clutch, and at least one inertially-loaded traction wheel connected to the friction clutch

that has a torsional compliance exhibiting an oscillatory response to torque inputs. The automatic clutch controller is preferably used with a transmission shift controller. This automatic clutch controller provides smooth clutch engagement during vehicle launch and following transmission shifts to minimize the oscillatory response to clutch engagement. This automatic clutch controller is useful in large trucks.

The automatic clutch controller receives inputs from an engine speed sensor and a transmission input speed sensor. The transmission input speed sensor senses the rotational speed at the input to the transmission, which is the output of the friction clutch. The automatic clutch controller develops a clutch engagement signal controlling a clutch actuator between fully disengaged and fully engaged positions. The clutch engagement signal engages the friction clutch in a manner causing asymptotic approach of the transmission input speed to a reference speed. This minimizes the oscillatory response to torque inputs of the inertially-loaded traction wheel.

In a launch mode, corresponding to normal start of the vehicle, or after transmission shifts, the clutch engagement signal causes the transmission input speed to asymptotically approach the engine speed.

The automatic clutch controller includes a PID regulator for insuring full clutch engagement within a predetermined interval of time after initial partial engagement. Any long term difference between the transmission input speed reference signal and the transmission input speed eventually drives the clutch to full engagement. The regulator bases proportional and integral control on the difference of engine speed and transmission input speed, while the differential term is also derived from that difference or from input speed alone. The PID regulator is relatively insensitive to inner loop variations and affords robust control characteristics which allow a given transmission to be employed on various types of large trucks and still retain good control characteristics.

The clutch controller includes construction to reduce the need for detailed particularization for individual vehicles or vehicle models. The PID regulator output is operated upon by a prefilter which serves to shape the system transient response. An algebraic summer forms the controlled error by subtracting the transmission input speed signal from the prefiltered signal. This error signal is supplied to a frequency compensator having sufficient gain as a function of frequency to reduce the system closed loop sensitivity to vehicle parameter variations thereby adding robustness to the control. The compensator produces a clutch engagement signal for controlling clutch engagement in a manner to minimize the oscillatory response to clutch engagement.

The automatic clutch controller is preferably implemented in discrete difference equations executed by a digital microcontroller. The microcontroller implements a compensator which reduces system error by increasing gain at low frequencies and has a transfer function approximately the inverse of the transfer function of the inertially-loaded traction wheel. This compensator transfer function includes a notch filter covering the region of expected oscillatory response of the driveline. The frequency band of this notch filter must be sufficiently broad to cover a range of frequencies because the oscillatory response frequency may change with changes in vehicle loading and driveline characteristics. The compensator also preferably provides an elevated response in the range of frequencies where the driveline response is a minimum to increase the loop gain and reduce sensitivity to variations in vehicle characteristics.

The clutch actuation controller preferably stores sets of coefficients for the discrete difference equations corresponding to each gear ratio of the transmission. The clutch actuation controller recalls the set of coefficients corresponding to the selected gear ratio. These recalled set of coefficients are employed in otherwise identical discrete difference equations for clutch control.

Brief Description of the Drawings

These and other objects and aspects of the present invention will be described below in conjunction with the drawings in which:

FIGURE 1 illustrates a schematic view of the vehicle drive train including the clutch actuation controller of the present invention;

FIGURE 2 illustrates the typical relationship between clutch engagement and clutch torque;

FIGURE 3 illustrates the ideal response of engine speed and transmission input speed over time for launch of the motor vehicle;

FIGURE 4 illustrates the function of an automatic clutch controller shown as a block diagram, according to the invention; and

FIGURE 5 is a schematic diagram of a PID regulator used in the controller of Figure 4.

Detailed Description of the Preferred Embodiments

Figure 1 illustrates in schematic form the drive train of a motor vehicle including the automatic clutch controller of the present invention. The motor vehicle includes engine 10 as a source of motive power. For a large truck of the type to which the present invention is most applicable, engine 10 would be a diesel internal combustion engine. Throttle 11, which is typically a foot operated pedal, controls operation of engine 10 via throttle filter 12. Throttle filter 12 filters the throttle signal supplied to engine 10 by supplying a ramped throttle signal up to the throttle level upon receipt of a step throttle increase via throttle 11. Engine 10 produces torque on engine shaft 15. Engine speed

sensor 13 detects the rotational speed of engine shaft 15. The actual site of rotational speed detection by engine speed sensor may be at the engine flywheel. Engine speed sensor 13 is preferably a multitooth wheel whose tooth rotation is detected by a magnetic sensor.

Friction clutch 20 includes fixed plate 21 and movable plate 23 that are capable of full or partial engagement. Fixed plate 21 may be embodied by the engine flywheel. Friction clutch 20 couples torque from engine shaft 15 to input shaft 25 corresponding to the degree of engagement between fixed plate 21 and movable plate 23. Note that while Figure 1 illustrates only a single pair of fixed and movable plates, those skilled in the art would realize that clutch 20 could include multiple pairs of such plates.

A typical torque verses clutch position function is illustrated in Figure 2. Clutch torque/position curve 80 is initially zero for a range of engagements before initial touch point 81. Clutch torque rises monotonically with increasing clutch engagement. In the example illustrated in Figure 2, clutch torque rises slowly at first and then more steeply until the maximum clutch torque is reached upon full engagement at point 82. The typical clutch design calls for the maximum clutch torque upon full engagement to be about 1.5 times the maximum engine torque. This ensures that clutch 20 can transfer the maximum torque produced by engine 10 without slipping.

Clutch actuator 27 is coupled to movable plate 23 for control of clutch 20 from disengagement through partial engagement to full engagement. Clutch actuator 27 may be an electrical, hydraulic or pneumatic actuator and may be position or pressure controlled. Clutch actuator 27 controls the degree of clutch engagement according to a clutch engagement signal from clutch actuation controller 60.

Transmission input speed sensor 31 senses the rotational velocity of input shaft 25, which is the input to transmission 30. Transmission 30 provides selectable drive ratios to drive shaft 35 under the control of transmission shift controller 33. Drive shaft 35 is coupled to differential 40. Transmission output speed sensor 37 senses the rotational velocity of drive shaft 35. Transmission input speed sensor 31 and transmission output speed sensor 37 are preferably constructed in the same manner as engine speed sensor 13 and provide directional sense as well as speed. In the preferred embodiment of the present invention, in which the motor vehicle is a large truck, differential 40 drives four axle shafts 41 to 44 that are in turn coupled to respective wheels 51 to 54.

Transmission shift controller 33 receives input signals from throttle 11, engine speed sensor 13, transmission input speed sensor 31 and transmission output speed sensor 37. Transmission shift controller 33 generates gear select signals for control of transmission 30 and clutch engage/disengage signals coupled to clutch actuation controller 60. Transmission shift controller 33 preferably changes the final gear ratio provided by transmission 30 corresponding to the throttle setting, engine speed, transmission input speed and transmission output speed. Transmission shift controller 33 provides respective engage and disengage signals to clutch actuation controller 60 depending on whether friction clutch 20 should be engaged or disengaged. Transmission shift controller also transmits a gear signal to clutch actuation controller 60. This gear signal permits recall of the set of coefficients corresponding to the selected gear. Note transmission shift controller 33 forms no part of the present invention and will not be further described.

Clutch actuation controller 60 provides a clutch engagement signal to clutch actuator 27 for controlling the position of movable plate 23. This controls the amount of torque transferred by clutch 20 according to clutch torque/position curve 80 of Figure 2. Clutch actuation controller 60 operates under the control of transmission shift controller 33. Clutch actuation controller 60 controls the movement of moving plate 23 from disengagement to at least partial engagement or full engagement upon receipt of the engage signal from transmission shift controller 33. In the preferred embodiment it is contemplated that the clutch engagement signal will indicate a desired clutch position. Clutch actuator 27 preferably includes a closed loop control system controlling movable plate 23 to this desired position. It is also feasible for the clutch engagement signal to represent a desired clutch pressure with clutch actuator 27 providing closed loop control to this desired pressure. Depending on the particular vehicle, it may be feasible for clutch actuator 27 to operate in an open loop fashion. The exact details of clutch actuator 27 are not crucial to this invention and will not be further discussed.

Clutch actuation controller 60 preferably generates a predetermined open loop clutch disengagement signal for a ramped out disengagement of clutch 20 upon receipt of the disengage signal from transmission shift controller 33. No adverse oscillatory responses are anticipated for this predetermined open loop disengagement of clutch 20.

Figure 3 illustrates engine speed 90 and transmission input shaft speed 100 for the case of launch, that is starting out from a stop in order to proceed at a reasonable speed. Initially, the engine speed 90 is at idle. Thereafter engine speed 90 monotonically increases within the time frame of Figure 3. Engine speed 90 either increases or remains the same. Ideally engine speed 90 increases until the torque produced by engine 10 matches the torque required to accelerate the vehicle. At high load this engine speed may be in the mid range between the idle speed and the maximum engine speed. This constant engine speed corresponds to the engine torque required to match clutch torque and driveline torque and achieve

a balance between engine output torque and the vehicle load torque. This torque level is the ideal clutch torque because a higher clutch torque would stall engine 10 and a lower clutch torque would allow the engine speed to increase too much. Ultimately the vehicle would accelerate to a speed where clutch 20 can be fully engaged. Thereafter the balance between engine torque and load torque is under the control of the driver via the throttle setting and clutch actuation controller 60 would continue to command full clutch engagement.

When the vehicle is stopped and clutch 20 fully disengaged, transmission input speed 100 is initially zero. This is the case for starting the vehicle. However, as further explained below, this same technique can be used for smooth clutch engagement upon shifting gears while moving. Thus the transmission input speed may initially be a value corresponding to the vehicle speed. Upon partial engagement of clutch 20, transmission input speed 100 increases and approaches engine speed 90 asymptotically. At a point 101, transmission input speed 100 is sufficiently close to engine speed 90 to achieve full engagement of clutch 20 without exciting the torsional compliance of the driveline of the vehicle. At this point clutch 20 is fully engaged. Thereafter transmission input speed 100 tracks engine speed 90 until clutch 20 is disengaged when the next higher final gear ratio is selected by transmission controller 33. The system preferably also operates for the case in which the vehicle is not stopped and the initial transmission input speed is nonzero.

Figure 4 illustrates schematically the control function of clutch actuation controller 60. As also illustrated in Figure 1, clutch actuation controller 60 receives the throttle signal from throttle 11, the engine speed signal from engine speed sensor 13 and the transmission input speed signal from transmission input speed sensor 31. Clutch actuation controller 60 illustrated in Figure 4 generates a clutch engagement signal that is supplied to clutch actuator 27 for operation of the friction clutch 20. Although not shown in Figure 4, the degree of clutch actuation, together with the throttle setting, the engine speed and the vehicle characteristics determine the transmission input speed that is sensed by transmission input speed sensor 31 and supplied to clutch actuation controller 60. Therefore, the control schematic illustrated in Figure 4 is a closed loop system.

The control function illustrated in Figure 4 is needed only for clutch positions between touch point 81 and full engagement. Clutch engagement less than that corresponding to touch point 81 provide no possibility of torque transfer because clutch 20 is fully disengaged. Clutch actuation controller 60 preferably includes some manner of detection of the clutch position corresponding to touch point 81. Techniques for this determination are known in the art. As an example only, the clutch position at touch point 81 can be determined by placing transmission 30 in neutral and advancing clutch 20 toward engagement until transmission input speed sensor 31 first detects rotation. Upon receipt of the engage signal from transmission shift controller 33, clutch actuation controller 60 preferably rapidly advances clutch 20 to a point corresponding to touch point 81. This sets the zero of the clutch engagement control at touch point 81. Thereafter the clutch engagement is controlled by the control function illustrated in Figure 4.

Clutch actuation controller 60 is preferably realized via a microcontroller circuit. Inputs corresponding to the engine speed, the transmission input speed and the throttle setting must be in digital form. These input signals are preferably sampled at a rate consistent with the rate of operation of the microcontroller and fast enough to provide the desired control. As previously described, the engine speed, transmission input speed and transmission output speed are preferably detected via multitooth wheels whose teeth rotation is detected by magnetic sensors. The pulse trains detected by the magnetic sensors are counted during predetermined intervals. The respective counts are directly proportional to the measured speed. For proper control the sign of the transmission input speed signal must be negative if the vehicle is moving backwards. Some manner of detecting the direction of rotation of input shaft 25 is needed. Such direction sensing is conventional and will not be further described. The throttle setting is preferably detected via an analog sensor such as a potentiometer. This analog throttle signal is digitized via an analog-to-digital converter for use by the microcontroller. The microcontroller executes the processes illustrated in Figure 4 by discrete difference equations in a manner known in the art. The control processes illustrated in Figure 4 should therefore be regarded as an indication of how to program the microcontroller embodying the invention rather than discrete hardware. It is feasible for the same microcontroller, if of sufficient capacity and properly programmed, to act as both clutch actuation controller 60 and as transmission shift controller 33. It is believed that an Intel 80C196 microcontroller has sufficient computation capacity to serve in this manner.

The engine speed is the reference signal for control; i.e. the engine speed is the desired transmission input speed. Clutch actuation controller 60 includes a PID (proportional-integral-differential) regulator 65, best shown in Figure 5. The transmission input speed from transmission input speed sensor 31 is subtracted from the engine speed in algebraic summer 64 to produce an error signal. The PID regulator 65 has an amplifier 84 coupled to the summer 64 output to set a gain and an integrator 86 integrates the amplified error signal which is fed to an algebraic summer 67. The error signal is also amplified by amplifier 88 to

provide a proportional term to the summer 67. The differential term to the summer 67 is supplied by amplifier 94 and differentiator 96 which operate on the transmission input speed signal. The proportional, integral and differential signals are added in the summer. Such regulators are well known in control methodology and may take different forms. For example, the differential term may be derived from the error signal as well as the proportional and integral terms.

Algebraic summer 67 supplies the input to prefilter 68. The output signal from prefilter 68 is supplied to algebraic summer 69. Algebraic summer 69 also receives the measured transmission input speed signal from transmission input speed sensor 31. Algebraic summer 69 forms the difference between the prefiltered signal from prefilter 68 and the transmission input speed. This difference is supplied to the compensator 70. The compensator 70 includes an approximate inverse model of the torsional oscillatory response of the vehicle to torque inputs. Compensator 70 includes a gain versus frequency function selected to reduce variations in the closed loop response of clutch actuation controller 60 due to variations in the transfer function of the vehicle driveline and in particular has increased gain at low frequencies to increase the robustness of the system. Determination of the transfer function of compensator 70 will be further described below. The compensator makes the inner loop predictable, having a small uncertainty, so that the system is controllable with the PID regulator 65 to achieve asymptotic approach of the input speed to the engine speed. The prefilter combines with the compensator to form a well-damped second order transfer function for control by the PID regulator. The character of prefilter 68 and its manner of determination will be further described below. The prefilter and compensator are treated separately in this description, but since they are series components they can be combined into the same unit; that is, both functions can be expressed as a single transfer function or the equivalent discrete difference equation or set of equations.

A feedforward signal is provided in the clutch engagement signal via an engine speed differential signal. Differentiator compensator or acceleration compensator 73 forms a differential signal responsive to the rate of change in the engine speed but is filtered to prevent abrupt decrease due to small engine deceleration. This engine speed differential signal and its integral formed by integrator 74 are supplied to algebraic summer 71. Algebraic summer 71 sums the output of compensator 70, the engine speed differential signal from acceleration compensator 73 and the integral signal from integrator 74 to form the clutch engagement signal. Clutch actuator 27 employs this clutch engagement signal to control the degree of clutch engagement.

The feedforward signal permits better response of clutch actuation controller 60 when the engine speed is accelerating. Under conditions of engine speed acceleration the feedforward signal causes rapid engagement of clutch 20 proportional to the rate of engine acceleration. The engine speed can increase rapidly under full throttle conditions before the driveline torque is established. This is because the speed of response of clutch actuation controller 60 without this feedforward response is low compared with the peak engine speed of response. With this feedforward response rapid engine acceleration results in more rapid than otherwise clutch engagement. The additional clutch engagement tends to restrain increase in engine speed by requiring additional torque from the engine. When the engine speed reaches a constant value, the differential term decays to zero and integrator 74 retains the clutch engagement needed to restrain engine speed. Other portions of the control function then serve to provide asymptotic convergence of the transmission input speed to the reference speed.

Prefilter 68 and compensator 70 perform differing and complementary functions in clutch actuation controller 60. The transfer functions of prefilter 68 and compensator 70 are determined as follows. The transfer function of compensator 70 is selected to reduce sensitivities of the closed loop transfer function to driveline parameter variations. This is achieved by providing sufficient loop gain as a function of frequency. If the sensitivity of the closed loop transfer function $H(\omega)$ with respect to the transfer function of the driveline $G(\omega)$ is $S^{H(\omega)}_{G(\omega)}$, then

$$S^{H(\omega)}_{G(\omega)} = \frac{1}{(1 + C(\omega)\,G(\omega))} \qquad (2)$$

where $C(\omega)$ is the transfer function of compensator 70. Inspection of this relationship reveals that the sensitivity $S^{H(\omega)}_{G(\omega)}$ can be reduced arbitrarily to zero by increasing the compensator gain. There are practical limits to the maximum compensator gain because of stability and noise problems. Thus the transfer function $C(\omega)$ of compensator 70 is selected high enough at all frequencies ω to limit the variations in the closed loop transfer function to an acceptable level set as a design criteria. Enhanced robustness is added by emphasizing the gain at low frequencies.

Compensator 70 includes an approximate inverse model of the torsional oscillatory response. In the typical heavy truck to which this invention is applicable, the torsional compliance of the driveline causes the driveline transfer function to have a pair of lightly damped poles that may range from 2 to 5 Hz. The exact value depends upon the vehicle parameter values. The inverse response of compensator 70 provides a notch filter in the region of these poles. The frequency band of the notch is sufficiently broad to cover the range of expected vehicle frequency responses. The typical heavy truck also includes a pair

of complex zeros in the frequency range from 1 to 2 Hz. These complex zeros tend to reduce the system loop gain and hence cause the system to be more sensitive to variations in vehicle characteristics in this frequency range. Compensator 70 preferably provides a pair of simple zeros in this frequency range to increase the loop gain and reduce sensitivity to variations in vehicle characteristics. Thus the total response of the closed loop system has highly damped eigen values providing a less oscillatory system.

Prefilter 68 is employed to reliably achieve a desired closed loop transient response. The transfer function $H(\omega)$ of the closed loop system without prefilter 68 is:

$$H(\omega) = \frac{C(\omega)\,G(\omega)}{(1 + C(\omega)\,G(\omega))} \quad (3)$$

where $C(\omega)$ is the transfer function of compensator 70 and $G(\omega)$ is the transfer function of the driveline. The above noted design for compensator 70 takes into account only reduction in sensitivity to variations in the driveline response $G(\omega)$. This typically results in a closed loop response $H(\omega)$ having an inappropriate time response. The design goal of the prefilter and compensator is to produce a predictable response characteristic thereby enabling the PID controller to achieve asymptotic convergence of the transmission input speed to engine speed through actuation of clutch 20. The transfer function $H(\omega)$ with prefilter 68 is:

$$H(\omega) = \frac{F(\omega)\,C(\omega)\,G(\omega)}{(1 + C(\omega)\,G(\omega))} \quad (4)$$

where $F(\omega)$ is the transfer function of prefilter 68. Prefilter 68 is a low pass filter with the pass band related to the design rate of the PID controller.

The above outlined determination of the response character of prefilter 68 and compensator 70 corresponds to the quantitative feedback theory of Horowitz. This theory is exemplified in "Quantitative Feedback Theory" by I. M. Horowitz, IEE Proceedings, Vol. 129, PT.d, no. 6, November 1982. This selection of the response of prefilter 68 and compensator 70 results in a system that is robust, that is, capable of properly responding to widely varying vehicle conditions.

As noted above, the elements of Figure 4 are preferably implemented via discrete difference equations in a microcontroller. Such equations have been specifically set forth in the above mentioned patent applications. The characteristics of the compensator and the prefilter may, however, be fully understood from the following expressions and discrete difference equations are readily derived by those skilled in the art. For the compensator, the transfer function is:

$$D(s) = \frac{6\,(s^2/35^2 + s/40 + 1)}{(s/7 + 1)\,(s/10 + 1)\,(s/15 + 1)\,(s/30 + 1)}. \quad (5)$$

Similarly the prefilter transfer function is:

$$F(s) = \frac{1}{(s/10 + 1)}. \quad (6)$$

The present invention can be advantageously employed for clutch re-engagement following shifts of the transmission. In this event the same control processes illustrated in Figure 4 would be employed, including the above listed transfer functions for prefilter 68 and compensator 70. When expressed as difference equations, the control processes for transmission shifts would differ from the launch process in selection of the coefficients used in the equations. A particular set of these coefficients would be recalled from coefficient memory 75 depending upon the gear signal from transmission shift controller 33. The selected set of coefficients may also include coefficients of integration for integrator 74, and coefficients for differentiator 73. In other respects the invention would operate the same as described above.

The control processes of the present invention are robust with regard to variations in vehicle response. It is believed that the automatic clutch controller herein described is capable of handling changes in vehicle loading within a single vehicle and variations in response between differing combinations of engine, clutch and driveline oscillatory response between different vehicles. Thus the automatic clutch controller of this invention need not be particularized for a particular vehicle. Thus the invention automatic clutch controller is easier to manufacture for a variety of vehicles.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

Claims

1. In a combination including a source of motive power (10) controlled by a throttle (11), a friction clutch (20) having an input shaft (15) connected to the source of the motive power (10) and an output shaft (25), and at least one inertially-loaded traction wheel (51, 52, 53, 54) connected to the output shaft (25) of the fraction clutch (20) having a torsional compliance exhibiting an oscillatory response to torque inputs, an automatic clutch controller comprising:

   an engine speed sensor (13) connected to the source of motive power (10) for generating an engine speed signal corresponding to the rotational speed of the source of motive power (10);

   a transmission input speed sensor (31) connected to the output shaft (25) of the friction clutch (20) for generating a transmission input speed signal corresponding to the rotational speed of the output shaft (25) of the friction clutch (20);

   a clutch actuator (27) connected to the friction clutch (20) for controlling engagement of the friction clutch (20) from disengaged to fully

engaged according to a clutch engagement signal; and

a controller (60) connected to said engine speed sensor (13), said transmission input speed sensor (31) and said clutch actuator (27) including

a PID regulator (65) coupled to the engine speed sensor (13) and the input speed sensor (31) for generated a regulated output,

a prefilter (68) connected to said PID regulator (65) for generating a filtered PID signal,

a first algebraic summer (69) connected to said transmission input speed sensor (31) and said prefilter (68) generating a first algebraic sum signal corresponding to the difference between (1) said filtered PID signal and (2) said transmission input speed signal, and

a compensator (70) connected to said first algebraic summer (69) for decreasing closed loop sensitivity and for generating said clutch engagement signal for supply to said clutch actuator (27) for engaging the friction clutch (20) in a manner causing said transmission input speed to asymptotically approach said engine speed.

2. The automatic clutch controller as claimed in claim 1, wherein the controller (60) connected to said engine speed sensor (13), said transmission input speed sensor (31) and said clutch actuator (27) includes:

means for determining the difference between the engine speed signal and the input speed signal; and

the PID regulator (65) operates on the said difference to generate the regulated output.

3. The automatic clutch controller as claimed in claim 1, wherein the controller (60) connected to said engine speed sensor (13), said transmission input speed sensor (31) and said clutch actuator (27) includes:

means for determining the difference between the engine speed signal and the input speed signal; and

the PID regulator (65) includes an integrator (86) having the said difference as its input to generate a component of the regulated output.

4. The automatic clutch controller as claimed in claim 1, further including:

a second algebraic summer (64) for determining the difference of the engine speed signal and the input speed signal;

the PID regulator (65) includes an integrator (86) coupled to the second algebraic summer (64) and responsive to the difference of the reference speed signal and the input signal for producing an integral term of the regulated output;

a differentiating compensator (73) responsive to the engine speed signal for producing a lead signal;

a second integrator (74) responsive to the lead signal for producing a second integrator signal; and

means for summing the output of the compensator (70), the lead signal and the second integrator signal to produce the clutch engagement signal.

5. In a combination including an engine (10) controlled by a throttle (11), a transmission (30) having an input shaft (25), a traction wheel (51, 52, 53, 54) driven by the transmission (30), and a friction clutch (20) connected between the engine (10) and the transmission input shaft (25) having a torsional compliance exhibiting an oscillatory response to torque inputs, and an automatic clutch control apparatus comprising sensors for the throttle position, engine speed, and input shaft speed, a clutch actuator (27) connected to the friction clutch (20) for controlling engagement of the friction clutch (20) from disengaged to fully engaged positions according to a clutch engagement signal; a method of generating a clutch engagement signal comprising the steps of:

determining the engine speed and the input shaft speed;

producing a proportional-integral-differential (PID) value from the engine speed and the input shaft speed;

filtering the PID value;

generating an error by subtracting the input shaft speed from the filtered PID value; and

compensating the error to produce a compensated output value by maintaining reduced sensitivity to variations in the response to torque inputs of the traction wheel to generate said clutch engagement signal to thereby engage the friction clutch (20) in a manner causing said transmission input speed signal to asymptotically approach said engine speed signal.

6. The method of generating a clutch engagement signal according to claim 5 comprising the further steps of:

forming an error value by subtracting the input shaft speed signal from the engine speed; and

the step of producing a PID value comprises producing a PID value from the error value.

7. The method of generating a clutch engagement signal according to claim 5 comprising the further steps of:

forming an error value by subtracting the input shaft speed signal from the engine speed; and

the step of producing a PID value includes generating an integral term and a proportional term from the error value.

8. The method of generating a clutch engagement signal according to claim 5 comprising the further steps of:

forming an error value by subtracting the input shaft speed signal from the engine speed; and

the step of producing a PID value comprises

generating an integral term and a proportional term from the error value,

generating a differential term from at least one of the input shaft speed and the engine speed, and

summing the integral, proportional and differential terms.

9. The method of generating a clutch engagement signal according to claim 5 comprising the further steps of:

differentiating the engine speed to produce a feedforward signal;

integrating the feedforward signal; and

combining the feedforward signal, the integrated feedforward signal and the compensated output value to generate said clutch engagement signal.

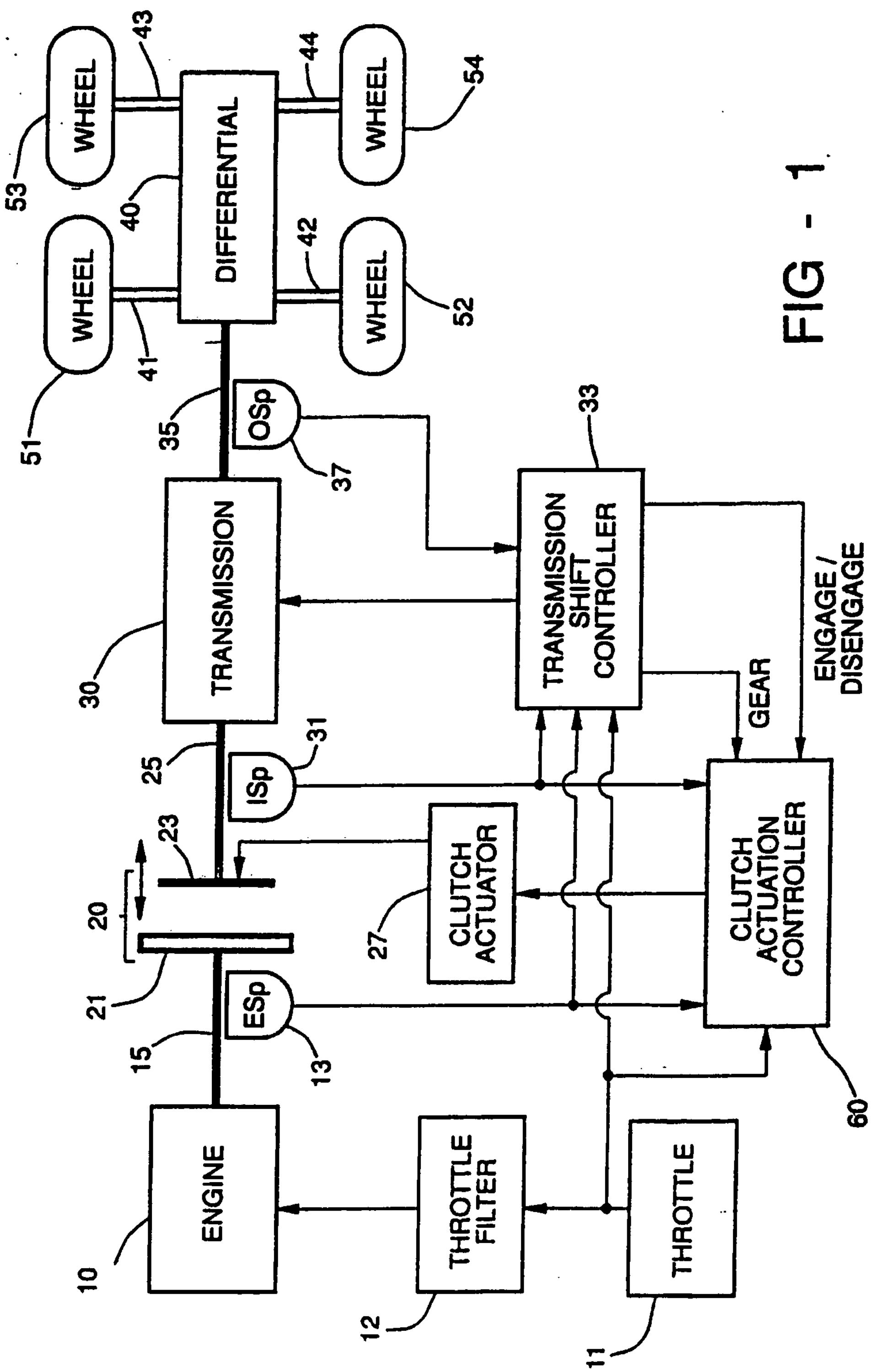

WHEEL
WHEEL
WHEEL
WHEEL
DIFFERENTIAL
TRANSMISSION
OSp
TRANSMISSION SHIFT CONTROLLER
ENGAGE / DISENGAGE
GEAR
ISp
CLUTCH ACTUATOR
CLUTCH ACTUATION CONTROLLER
ESp
ENGINE
THROTTLE FILTER
THROTTLE
10
11
12
13
15
20
21
23
25
27
30
31
33
35
37
40
41
42
43
44
51
52
53
54
60

FIG - 1

FIG - 2

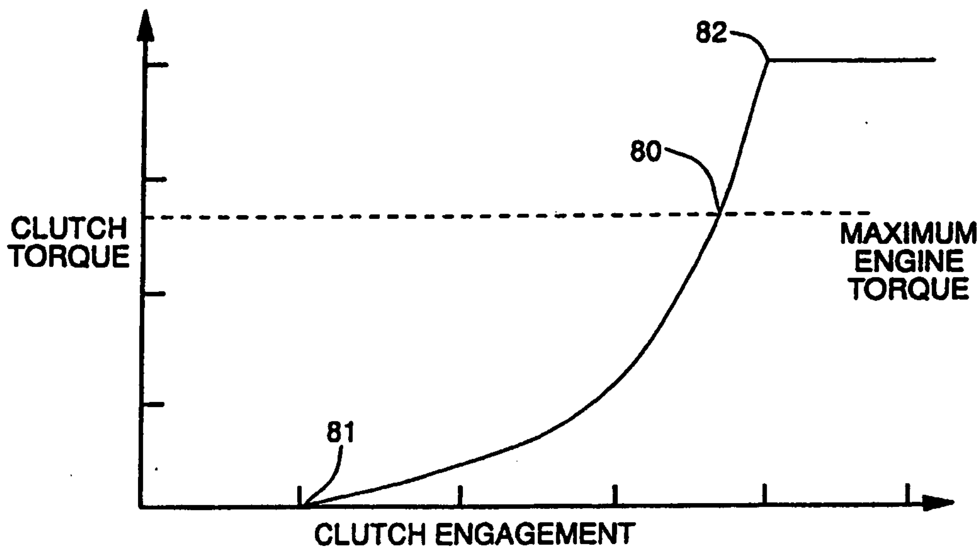
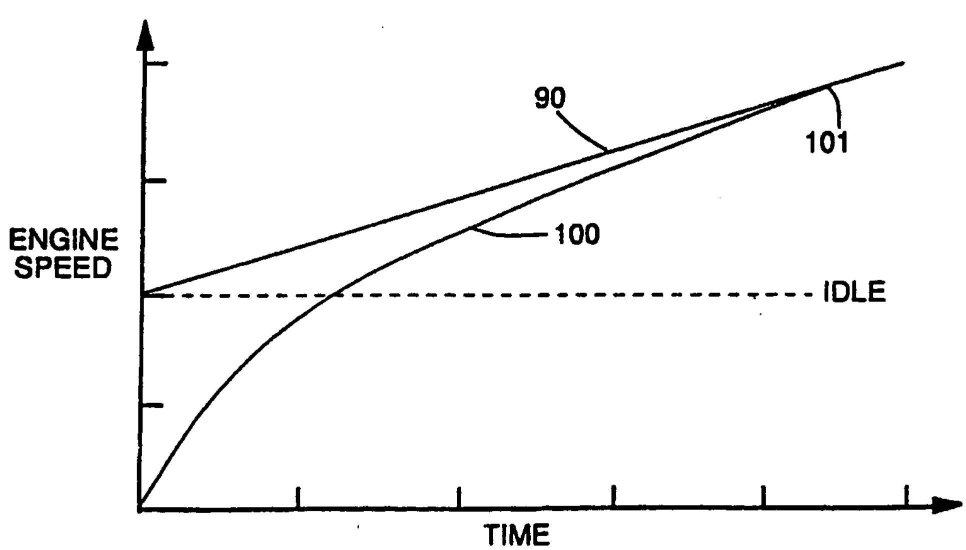

82
80
CLUTCH
TORQUE
MAXIMUM
ENGINE
TORQUE
81
CLUTCH ENGAGEMENT

FIG - 3


90
101
ENGINE
SPEED
100
IDLE
TIME

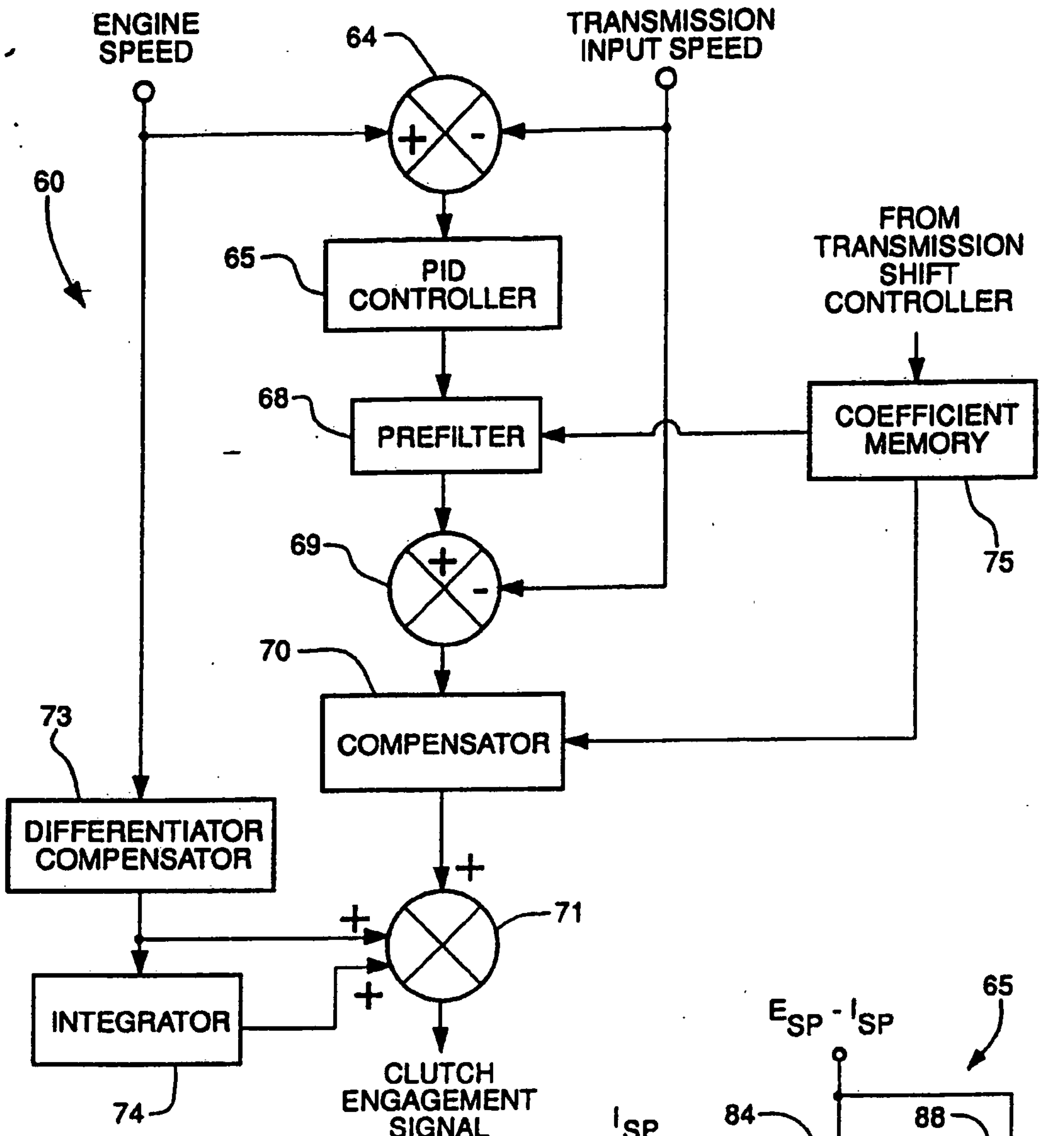

ENGINE SPEED
64
TRANSMISSION INPUT SPEED
60
65
PID CONTROLLER
FROM TRANSMISSION SHIFT CONTROLLER
68
PREFILTER
COEFFICIENT MEMORY
75
69
70
COMPENSATOR
73
DIFFERENTIATOR COMPENSATOR
71
INTEGRATOR
74
CLUTCH ENGAGEMENT SIGNAL

FIG - 4

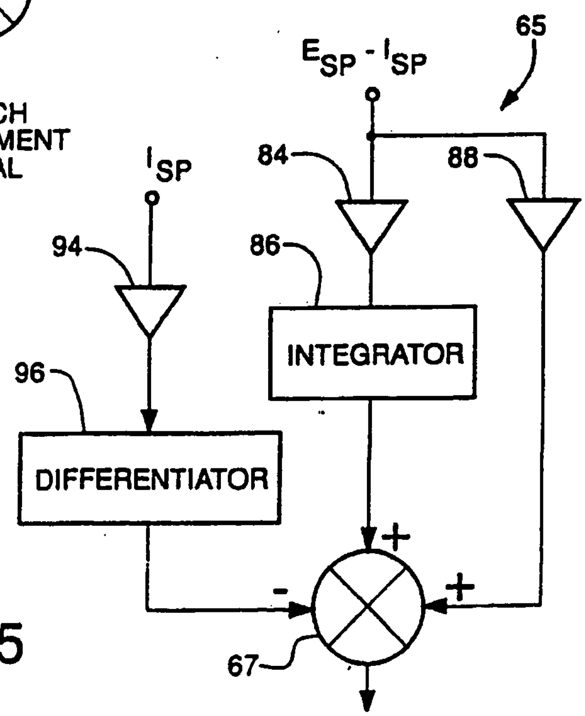

65
$E_{SP} - I_{SP}$
I_{SP}
84
88
94
86
INTEGRATOR
96
DIFFERENTIATOR
67

FIG - 5

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 95 30 0886

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
D,X	US-A-5 275 267 (SLICKER) * the whole document * ---	1-9	F16D48/06
A	EP-A-0 494 608 (FICHTEL & SACHS AG) * the whole document * -----	1-9	

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
F16D
B60K

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	6 June 1995	Clasen, M

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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